Further development on the LPV fault tolerant control for vehicle dynamics

Soheib Fergani, Olivier Sename, Luc Dugard, Peter Gáspár

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S. Fergani[1*], O. Sename[1], L. Dugard[1], P. Gáspár[2]

[1] *GIPSA-lab, Control Systems Dept, CNRS-Grenoble INP, ENSE3, BP 46, F-38402 St Martin d'Hères cedex, France (e-mail: {soheib.fergani, olivier.sename, luc.dugard}@gipsa-lab.grenoble-inp.fr).*
[2] *Computer and Automation Research Institute, Hungarian Academy of Sciences, Kende u. 13-17, H-1111 Budapest, Hungary.*

Abstract: This paper aims at presenting the efficiency of the Linear Parameter Varying method for vehicle dynamics control, in particular when some actuators may be in failure. The case of the suspension actuators failure and braking actuators failure are presented.
The main objective is to enhance the vehicle dynamics even with faulty actuators through the suspension control (for comfort and road holding improvements) and the braking/steering control (for road handling and safety).
Indeed, the LPV/H_∞ fault tolerant MIMO gain-scheduled Vehicle Dynamic Control (VDSC) involves the steering actuators, rear brakes and four suspension systems, and aims at enhancing the yaw stability, lateral and vertical car performances (see Poussot-Vassal et al. (2011b), Doumiati et al. (2013)).
This strategy is scheduled by 3 varying parameters (ρ_b, ρ_s and ρ_l). These parameters depend on a special monitoring system defined to evaluate the impact of braking/suspension actuator failures on the vehicle dynamical performances.
The proposed LPV control structure then allows to handle such failures by an online adaptation of the control input distribution.
Simulation results performed on a nonlinear model experimentally validated on a vehicle Renault Mégane Coupé MIPS (Mulhouse) subject to critical driving situations show that the proposed methodology is effective and robust.

Keywords: Vehicle dynamics, Braking, Suspension, Steering,load transfer distribution, LPV, Fault Tolerant control, $\mathcal{H}_\infty$ control.

1. INTRODUCTION

Automotive light vehicles are complex systems involving different dynamics. Nowadays, automotive systems use more and more actuators and sensors. The 3 main subsystems that influence vehicle dynamics are suspensions for the vertical displacements while lateral and longitudinal dynamics mainly depend on braking and steering systems. In this context an important problem is the communication and coordination between those systems. A lot of studies have proposed several strategies for the vehicle dynamics control (see Kiencke and Nielsen (2000), Milliken and Milliken (1995), Gillespie (1992)). Indeed, the increasing number of actuators and sensors gives more degrees of freedom of the vehicle dynamics control, but also may cause some issues in the case of equipment failures. The risks of accident is then very high.
FTC (Fault Tolerant Control) main objective is to keep the normal operation system when some malfunctions and/or failures appear (see Blanke et al. (1997)). Then, this kind of control aims at ensuring the closed-loop system stability, and some level of performance, which could be degraded. The most intuitive method is the physical redundancy with duplication of actuator and sensor components, but due to the cost increase, analytical redudancy is often preferred, trough estimation and control algorithms, designed to handle the subsystem malfunction.
In this study, a new multivariable fault tolerant LPV/H_∞ Global Chassis control strategy is proposed to manage actuator failures. It also allows to achieve several performance objectives using a smart control structure that adapts the control to the considered vehicle dynamical behaviour. The interest of this control allocation like approach is to provide the ad-hoc control inputs distribution that achieves the desired objectives. In addition the LPV control structure allows to simplify the implement step, avoiding the use of a bank of several controllers.
Indeed, here the proposed strategy in addition of achieving the damper failure management allows to handle the braking malfunction to avoid critical driving within global chassis control strategy as shown in the scheme in Fig. 1: Authors have already provided some solutions to treat the case of one damper failure in some driving situations (see Sename et al. (2013). Indeed, the allocation control strategy was used to provide the accurate suspension effort in each corner to maintain the vehicle stability when one of the dampers is faulty.
In this paper, an extension to this works is provided

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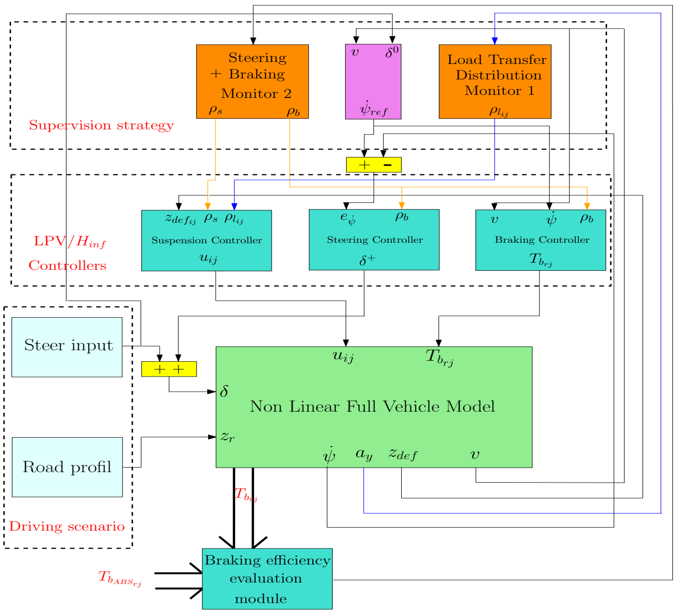

Fig. 1. Global chassis control implementation scheme.

including several actuator failures simultaneously in critical driving situations. The LPV framework is achieved thanks to some varying parameters that allow to adapt the control structure regarding the performance objective to be achieved.Simulation results using a critical driving scenario prove the efficiency of the proposed strategy.
The paper is structured as follows: Section 2 the driving situation monitoring approach and the varying parameter generation is introduced. Section 3 presents the propose new global chassis control strategy. .In Section 4, the simulation results that prove the efficiency of the proposed global chassis fault tolerant strategy are given.

Notations and vehicle parameters:
Throughout the paper, the following notation will be adopted: indices $i = \{f, r\}$ and $j = \{l, r\}$ are used to identify vehicle front, rear and left, right positions respectively. Then, index $\{s, t\}$ holds for forces provided by suspensions and tires respectively. $\{x, y, z\}$ holds for forces and dynamics in the longitudinal, lateral and vertical axes respectively. Then let $v = \sqrt{v_x^2 + v_y^2}$ denote the vehicle speed, $R_{ij} = R - (z_{us_{ij}} - z_{r_{ij}})$ the effective tire radius, $m = m_s + m_{us_{fl}} + m_{us_{fr}} + m_{us_{rl}} + m_{us_{rr}}$ the total vehicle mass, $\delta = \delta_d + \delta^+$ is the steering angle (δ_d, the driver steering input and δ^+, the additional steering angle provided by steering actuator, and $T_{b_{ij}}$ the braking torque provided by the braking actuator (see table.1). The model parameters, are those of a Renault Mégane Coupé (see Poussot-Vassal et al. (2011a)), obtained during a collaborative study with the MIPS laboratory in Mulhouse, through identification with the real data.

2. DRIVING SITUATION SUPERVISION AND SCHEDULING PARAMETERS GENERATION

Since attitude and yaw stability are concerned in this study, the strategy based on the measurement of the longitudinal slip ratio (see Poussot-Vassal et al. (2011a)) of the rear wheels (s_{rj}) is efficient while being simple. Both scheduling parameters are defined as follows:

Symbol	Value	Unit	Signification
m_s	350	kg	suspended mass
$m_{us_{fj}}$	35	kg	front unsprung mass
$m_{us_{rj}}$	32.5	kg	rear unsprung mass
$I_x; I_y; I_z$	250; 1400; 2149	$kg.m^2$	roll, pitch, yaw inertia
I_w	1	$kg.m^2$	wheel inertia
$t_f; t_r$	1.4; 1.4	m	front, rear axle
$l_f; l_r$	1.4; 1	m	COG-front, rear distance
R	0.3	m	nominal wheel radius
h	0.4	m	chassis height

Table 1. Renault Mégane Coupé parameters

(1) **Monitor on the braking efficiency:** The aim of the monitor is to schedule the GCC control to activate the steering system when braking is no longer efficient enough to guarantee safety. Then, one proposes the following scheduling strategy:

$$\rho_b = \max(|e_{T_{b_{rj}}}|) \ , \ j = \{l, r\} \tag{1}$$

where $e_{T_{b_{rj}}} = T_{b_{ABS_{rj}}} - T^*_{b_{rj}}$, and one defines the scheduling parameter $\xi(e)$ as:

$$\rho_b := \begin{cases} \overline{\xi} & \text{if } \rho_b \leq \underline{\chi} \\ \dfrac{\overline{\chi} - e}{\overline{\chi} - \underline{\chi}}\overline{\xi} + \dfrac{e - \underline{\chi}}{\overline{\chi} - \underline{\chi}}\underline{\xi} & \text{if } \underline{\chi} < \rho_b < \overline{\chi} \\ \underline{\xi} & \text{if } \rho_b \geq \overline{\chi} \end{cases} \tag{2}$$

where $\underline{\chi} = \frac{30}{100} T_{b_{max}}$ and $\overline{\chi} = \frac{70}{100} T_{b_{max}}$ are user defined brake efficiency measures. Note that other monitor strategies may be employed.

(2) **Suspension and Steering monitor according to the braking efficiency:** ρ_s is defined as :

$$\rho_s \begin{cases} \to 1 & \text{when } 1 > \rho_b > R^2_{crit} \\ = \dfrac{\rho_b - R^1_{crit}}{R^2_{crit} - R^1_{crit}} & \text{when } R^1_{crit} < \rho_b < R^2_{crit} \\ \to 0 & \text{when } 0 < \rho_b < R^1_{crit} \end{cases} \tag{3}$$

when $\rho_b > R^2_{crit} (= 0.9)$, i.e. when a low slip ($< s^-$) is detected, the vehicle is not in an emergency situation and ρ_s is set to 0. When $\rho_b < R^1_{crit} (= 0.7)$, i.e. when a high slip occurs ($> s^+$), a critical situation is reached and ρ_s is set to 0. Intermediate values of ρ_b will give intermediate driving situations.

(3) **The suspension control distribution for the dampers malfunction management**: the ρ_l is used to generate the adequate suspension forces in the four corners of the vehicle depending on the load transfer (left ⇆ right) caused by the performed driving scenario. It is based on the evaluation of the roll load transfer when the vehicle is running in several situations. The main idea is to compute the difference between the right and left vertical forces at the four corners of the vehicle. This suspension monitor is characterized by the following equations:

$$\begin{cases} F_{z_l} &= m_s \times g/2 + m_s \times h \times a_y/l_f \\ F_{z_r} &= m_s \times g/2 - m_s \times h \times a_y/l_r \\ \rho_l &= (F_{z_l} - F_{z_r})/(F_{z_l} + F_{z_r}); \end{cases} \tag{4}$$

where, F_{z_l} and F_{z_r} are the vertical forces, a_y the lateral acceleration, ρ_l the scheduling parameter. Note that $\rho_l \in [-1 \ \ 1]$.

Remark 1. The controllers are derived thanks to LPV/$\mathcal{H}_\infty$ methodology. This framework allows to smoothly tune the control performances thanks to the scheduling parameters ρ_b et ρ_s, guaranteeing internal stability (avoiding switching) and ensuring $\mathcal{H}_\infty$ performances.

Then, the LPV/$\mathcal{H}_\infty$ FT GCC strategy focuses on achieving in the same control structure the following objectives:

- **The suspension control reconfiguration under damper malfunction**: this fault tolerant control strategy which handles vehicle roll dynamics under damper malfunction (see Sename et al. (2013) and Fergani et al. (2014)) proposes a new structure of the controller, by making the corresponding LMIs orthogonal with a parameter dependency on the controller matrix output (see Savaresi et al. (2010), Fergani et al. (2013b) and Fergani et al. (2013a)).. It is worth noting that for this part of the study active suspensions are considered in the control and simulation procedures.
- **The control adaptation to critical driving situations and braking actuator malfunction**: Two scheduling parameters ρ_b and ρ_s will be used to coordinate the actuators and provide hierarchical use of the 3 VDSC actions (steering, braking and active suspension). When dangerous situation is detected, the GCC gives a torque reference to the braking system (that avoids slipping thanks to the ABS local controller), and if the braking system is not efficient enough and is not able to stabilize the vehicle (e.g. in case of low adherence or braking failure), the steering system is activated, and the suspension performances are changed from soft to hard, in order to handle the dynamical problem.

3. GLOBAL CHASSIS CONTROL DESIGN STRATEGY

The synthesis of the different controllers is completed in 2 steps, to decouple lateral and vertical dynamics. The coupling effects are handled through the scheduling parameter ρ_s and thanks to an "anti-roll" action of the suspension systems.

- First the steering/braking controllers are designed using the linear bicycle model, to improve the lateral dynamics and to stabilize the vehicle.
- Then the suspension controllers are synthesized, using the linear vertical full car model, to improve the comfort/road handling performance objectives and the vertical dynamics behavior.

Below, LPV/$\mathcal{H}_\infty$ controllers (with ρ_b and ρ_s the scheduling parameters) are developed thanks to a dedicated polytopic approach (for more details, see Scherer (1996)).

3.1 Step1: the braking/steering control Problem formulation

For the the braking/steering controller design, the following extended bicycle model is used.

This model emphasises the lateral dynamics of the vehicle. It is used especially for the design of the steering and braking controllers. The corresponding dynamical equations are in Eq. 5. The considered LPV/$\mathcal{H}_\infty$ control problem is described in Fig. (2) with the following scheduled weighting functions:

- $W_{e_{\dot\psi}} = 10\frac{s/500+1}{s/50+1}$, is used to shape the yaw rate error
- $W_{\dot v_y} = 10^{-3}$, attenuates the lateral acceleration
- $W_{T_{b_{rj}}}(\rho_b) = (1-\rho_b)\frac{s/10\varpi+1}{s/100\varpi+1}$, attenuates the yaw moment control input
- $W_{\delta^+}(\rho_s) = \rho_s\frac{s/\kappa+1}{s/10\kappa+1}$, attenuates the steering control input according to the value of ρ_s

where ϖ (resp.κ) is the braking (resp.steering) actuator cut-off frequency.

- When the tire force is in the linear zone, i.e. there is no risk of locking; so $\rho_b \to 1$ and the weighting function gain of $W_{T_{b_{rj}}}$ is chosen to be low. Therefore, the braking control is activated.
- When a high slip ratio is detected (critical situation) , the tire may lock, so $\rho_b \to 0$ and the gain of the weighting function is set to be high. This allows to deactivate the braking signal leading to a natural stabilisation of the slip dynamic.

On the other hand, when the driving situation is dangerous and presents a high risk for passengers, the steering control is activated through $W_{\delta^+}(\rho_s)$. The steering action depends on the varying parameter ρ_s, with $\rho_s(.) \in \mathcal{P}_{\rho_s}$ and $\mathcal{P}_{\rho_s} := \{\rho_s \in \mathbb{R} : \underline{\rho_s} \le \rho_s \le \overline{\rho_s}\}$ (where $\underline{\rho_s} = 0.1$ and $\overline{\rho_s} = 1$).

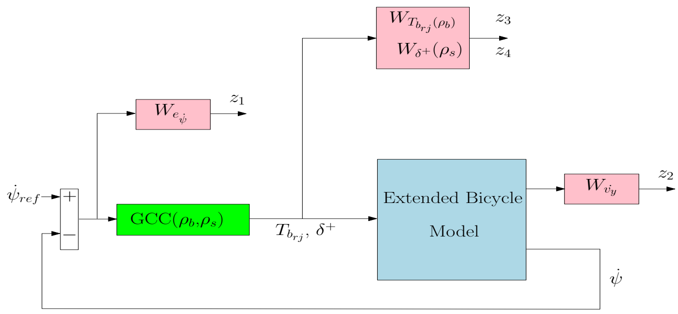

Fig. 2. Generalized plant for braking/steering control synthesis.

The generalized plant corresponding to Fig. 2 is LPV and can be modeled as,

$$\Sigma(R(.)) : \begin{bmatrix} \dot x \\ z \\ y \end{bmatrix} = \left[\begin{array}{c|cc} A(\rho(.)) & B_1(\rho(.)) & B_2 \\ C_1(\rho(.)) & D_{11}(\rho(.)) & D_{12} \\ C_2 & 0 & 0 \end{array}\right] \begin{bmatrix} x \\ w \\ u \end{bmatrix} \quad (6)$$

where x includes the state variables of the system and of the weighing functions, $w = F_{dy}$ and $u = [\delta^+, T_{b_{rj}}]$ are the exogenous and control inputs respectively.
$z = [z_1, z_2, z_3, z_4]$
$= [W_{e_{\dot\psi}} e_{\dot\psi}, W_{\dot v_y}\dot v_y, W_{T_{b_{rj}}}(\rho_b)T_{b_{rj}}, W_{\delta^+}(\rho_s)\delta^+]$ holds for the controlled output, and $y = \dot\psi_{ref}(v) - \dot\psi$ is the controller input ($\dot\psi_{ref}(v)$ is provided by a reference bicycle model).

Notice that the LPV model (6) is affine w.r.t parameters ρ_s and ρ_b and can be described as a polytopic system, i.e. a convex combination of the systems defined at each vertex formed by $\mathcal{P}_\rho(.)$, namely $\Sigma(\underline{\rho}(.))$ and $\Sigma(\overline{\rho}(.))$.

3.2 Step 2: the suspension control problem formulation

For this controller design, a 7 DOF vehicle model is considered.

$$\begin{bmatrix} \dot{v}_y \\ \dot{\psi} \\ \dot{\beta} \end{bmatrix} = \begin{bmatrix} \frac{-C_f - C_r}{mv} & v - \frac{-C_f l_f + C_r l_r}{mv} & 0 \\ \frac{-C_f l_f + C_r l_r}{I_z v} & \frac{-C_f l_f^2 - C_r l_r^2}{I_z v} & 0 \\ 0 & 1 + \frac{l_r C_r - l_f C_f}{mv^2} & -\frac{C_f + C_r}{mv} \end{bmatrix} \begin{bmatrix} v_y \\ \psi \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{C_f}{m} & -\frac{1}{m} & 0 \\ \frac{C_f l_f}{I_z} & 0 & \frac{t_r}{R I_z} \\ \frac{C_f}{mv} & 0 & \frac{1}{mv} \end{bmatrix} \begin{bmatrix} \delta \\ F_{dy} \\ T_{b_{rj}} \end{bmatrix} \tag{5}$$

This model includes the vertical dynamics of the chassis, the vertical motions of the wheels and the pitch and roll, respectively, z_s, $z_{us_{ij}}$, θ,and ϕ. The dynamical equations are in Eq. 7.

$$\begin{cases} \ddot{z}_s = -\left(F_{sz_f} + F_{sz_r} + F_{dz}\right)/m_s \\ \ddot{z}_{us_{ij}} = \left(F_{sz_{ij}} - F_{tz_{ij}}\right)/m_{us_{ij}} \\ \ddot{\theta} = \left((F_{sz_{rl}} - F_{sz_{rr}})t_r + (F_{sz_{fl}} - F_{sz_{fr}})t_f + mh\dot{v}_y\right)/I_x \\ \ddot{\phi} = \left(F_{sz_f} l_f - F_{sz_r} l_r - mh\dot{v}_x\right)/I_y \end{cases} \tag{7}$$

where $F_{tz_i} = F_{tz_{il}} + F_{tz_{ir}}$ and $F_{sz_i} = F_{sz_{il}} + F_{sz_{ir}}$, stand for the vertical tire forces and the suspension forces respectively. Index $i = \{f, r\}$ and $j = \{l, r\}$ are used to identify vehicle front, rear and left, right positions respectively. This model is mainly used for control design purposes. it provides information on the vertical dynamics of the car.
For the control design purposes, linear models are assumed for the stiffness k_{ij} and damping c_{ij} in the suspension force computation.
In this step, the suspension control with performance adaptation (see Savaresi et al. (2010)), to be integrated in the global VDSC strategy (Vehicle Dynamic Control), is presented. The following H_∞ control scheme is considered, including parameter varying weighting functions.
Indeed, the following LPV/$\mathcal{H}_\infty$ FT control design scheme is considered, including parameter varying weighting functions. where $W_{z_s} = \rho_s \frac{s^2 + 2\xi_{11}\Omega_{11}s + \Omega_{11}{}^2}{s^2 + 2\xi_{12}\Omega_{12}s + \Omega_{12}{}^2}$ is shaped in order

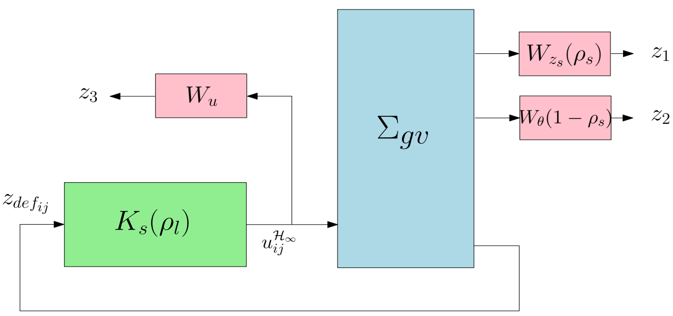

Fig. 3. Suspension system generalized plant.

to reduce the bounce amplification of the suspended mass (z_s) between $[0, 12]$Hz.
$W_\theta = (1 - \rho_s)\frac{s^2 + 2\xi_{21}\Omega_{21}s + \Omega_{21}{}^2}{s^2 + 2\xi_{22}\Omega_{22}s + \Omega_{22}{}^2}$ attenuates the roll bounce amplification in low frequencies.
$W_u = 3.10^{-2}$ shapes the control signal.

Remark 2. The parameters of these weighting functions are obtained using genetic algorithm optimization as in Do et al. (2010).

According to Fig. 3, the following parameter dependent suspension generalized plant ($\Sigma_{gv}(\rho_s)$) is obtained:

$$\Sigma_{gv}(\rho_s, \rho_l) := \begin{cases} \dot{\xi} = A(\rho_s, \rho_l)\xi + B_1\tilde{w} + B_2 u \\ \tilde{z} = C_1(\rho_s, \rho_l)\xi + D_{11}\tilde{w} + D_{12}u \\ y = C_2\xi + D_{21}\tilde{w} + D_{22}u \end{cases} \tag{8}$$

where $\xi = [\chi_{vert} \ \ \chi_w]^T$; $\tilde{z} = [z_1 \ \ z_2 \ \ z_3]^T$; $\tilde{w} = [z_{rij} \ \ F_{dx,y,z} \ \ M_{dx,y}]^T$; $y = z_{def_{ij}}$; $u = u_{ij}^{\mathcal{H}_\infty}$; and χ_w are the vertical weighting functions states.
As in previously defined,the parameter ρ_l to schedule the distribution of the left & right suspensions on the four corners of the vehicle and tune the suspension dampers smoothly, thanks to the LPV frame work, from "soft" to "hard" to improve the car performances according to the driving situation. This distribution is handled using a specific structure of the suspension controller, given as follows :

$$K_s(\rho) := \begin{cases} \dot{x}_c(t) = A_c(\rho_s)x_c(t) + B_c(\rho_s)y(t) \\ \begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \underbrace{U(\rho_l)C_c^0(\rho_s)}_{C_c(\rho_s, \rho_l)} x_c(t) \end{cases} \tag{9}$$

where $x_c(t)$ is the controller state, $A_c(\rho_s)$, $B_c(\rho_s)$ and $C_c(\rho_s)$ controller scheduled by ρ_s.
$u^{\mathcal{H}_\infty}(t) = [u_{fl}^{\mathcal{H}_\infty}(t)u_{fr}^{\mathcal{H}_\infty}(t)u_{rl}^{\mathcal{H}_\infty}(t)u_{rr}^{\mathcal{H}_\infty}(t)]$ the input control of the suspension actuators and $y(t) = z_{def}(t)$.

In this synthesis, the authors wish to stress that an interesting innovation is the use of a partly fixed structure controller, combined with a parameter dependency on the control output matrix introduced to allow a smooth load transfer distribution, depending on the situation. Then, the LPV framework is obtained, thanks to the matrix $U(\rho_l)$,

$$U(\rho_l) = \begin{pmatrix} 1 - \rho_l & 0 & 0 & 0 \\ 0 & \rho_l & 0 & 0 \\ 0 & 0 & 1 - \rho_l & 0 \\ 0 & 0 & 0 & \rho_l \end{pmatrix} \tag{10}$$

It is worth noting that this control design structure allows to tune various actuators controllers depending on the driving situation, by a hierarchical activation to optimize the use of them(coordinate framework with smooth transition between different performance objectives even if they are contradictory.

4. SIMULATION RESULTS

To test the efficiency of the proposed strategy, the following driving scenario is considered, including a braking actuator fault and a damper fault: the vehicle runs at 100km/h on a wet road ($\mu = 0.5$) in straight line. Then, two 5cm Road bump occurs from $t = 0.5$s to $t = 1.5$s and from $t = 4$s to $t = 5$s). After, A double line change manoeuvre is performed (from $t = 2$s to $t = 6$s) by the driver. Also, two types of faults are considered: a saturation of $75N$ on the left rear braking actuator is applied to simulate the fault on the braking system at the beginning of the line change and a fault on front left damper (force limitation of 70% occurs at $t = 4$s). Lateral wind occurs at vehicle's front generating an undesirable yaw moment (from $t = 2.5$s to $t = 3$s).

The resulting monitoring signals ρ_b and ρ_s and ρ_l are obtained (see Fig. 6). This parameters allow to achieve the online adaptation of the proposed control structure to the performed driving scenarios and situations. These

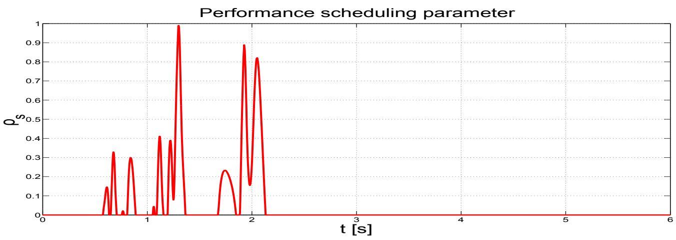

Fig. 4. Steering/suspension scheduling parameter ρ_s

parameters allow to activate or deactivate the control actions, when required. Note that ρ_b monitors the braking efficiency (compared to an ABS system). These parameters

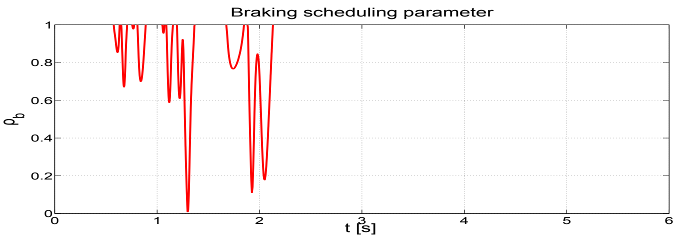

Fig. 5. Braking monitoring parameter ρ_b

allow to activate or deactivate the control actions, when required. Note that ρ_b monitors the braking efficiency (compared to an ABS system).
The ρ_s scheduling parameter, depends on the value of ρ_b. It also provides the necessary assistance to the driver by giving an additional steering δ^+ and setting the suspension dampers to "hard" to enhance road handling in critical situations. Also ρ_l allows to distribute the suspensions

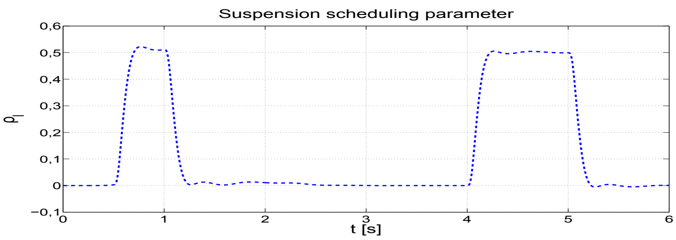

Fig. 6. Control allocation scheduling parameter ρ_l

efforts depending on load transfer left/right to manage the overload on each corner of the vehicle by generating the adequate efforts. It can be seen from Fig. 7 that the proposed strategy enhances the vehicle lateral stability. The vehicle yaw rate is considerably enhanced by the LPV approach, which improve very well car lateral dynamics.

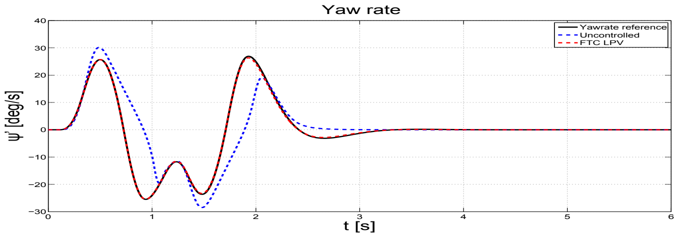

Fig. 7. Yaw rate

Remark 3. For Fig. 7, a "reference vehicle" yaw rate is given to have a better idea on the improvement brought by the proposed LPV strategy.

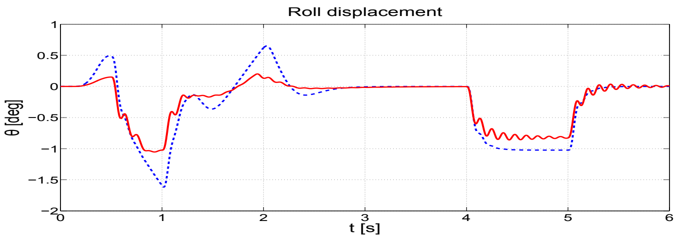

Fig. 8. Roll motion of the chassis

Fig. 8 shows that the LPV design strategy, in addition of enhancing vehicle stability, improves the vertical dynamics. It can be seen that the roll dynamics are considerably attenuated which enhance the vehicle handling when facing critical driving situations. Fig. 9 summarizes

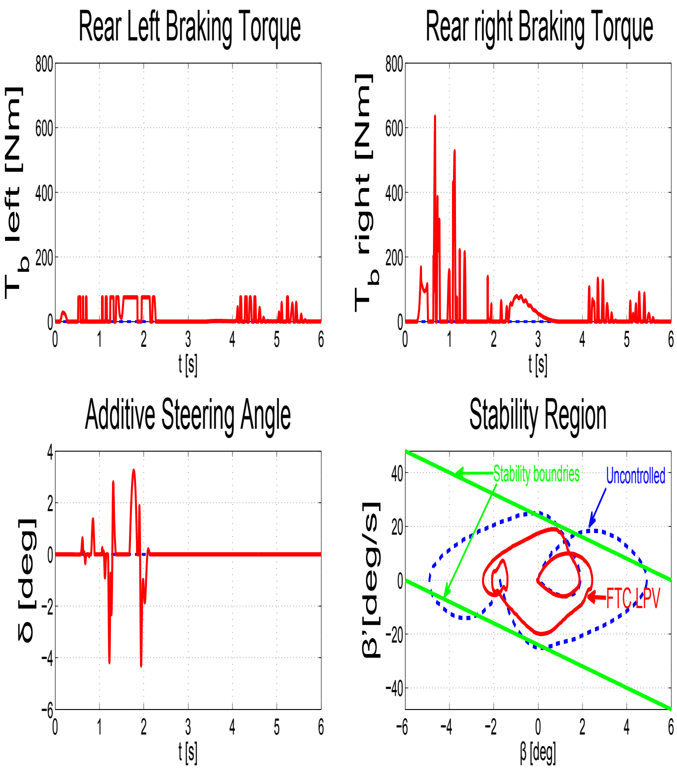

Fig. 9. additive steering input, braking Actuators torques and the vehicle stability evaluation

the braking/steering actuators actions and their effects on the vehicle stability. Indeed, it can be seen that the rear left braking actuators is faulty and its effort saturates quickly at low value (75 Nm) which simulate the actuator

failure and generate a instability risk. Then, the remanning healthy braking actuators provides more effort to compensate the lack of the braking torque of the faulty actuator and also the steering control is activated to help keeping the vehicle stability. It is worth to note that the proposed LPV control design structure avoids the actuators saturation while coordinating hierarchically their work.
The last result in Fig. 9 shows the efficiency of the proposed strategy in term of vehicle stabilization. It can be clearly seen that the good coordination of the vehicle steering, braking and suspension improves very well the vehicle behaviour and enhance the various car dynamics (vertical, lateral...). The vehicle is kept, by the proposed LPV/$\mathcal{H}_\infty$, from going beyond the limits of the stability region (based on the sideslip stability observation of the vehicle) even when performing a dangerous driving situation. Fig. 10 shows the result of the suspension control

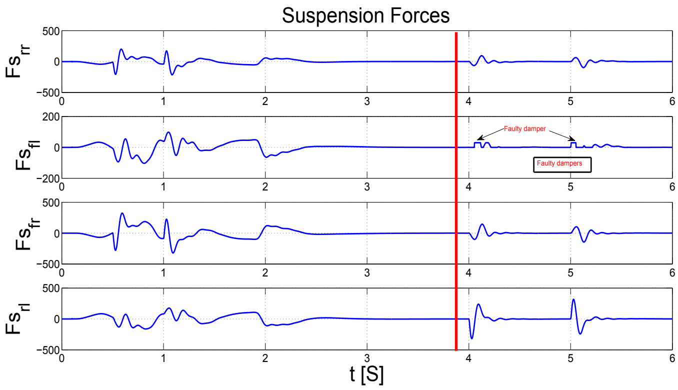

Fig. 10. Suspension dampers efforts

allocation strategy with the considered faulty dampers. Indeed, it can be seen that one of the dampers (the front left damper) provides a lower effort and is quickly saturated. The proposed LPV/$\mathcal{H}_\infty$ fault tolerant control strategy allows to manage this actuators failure by reconfiguration the suspension control to compensate the lack of the damping force in of the vehicle corners. This aims at ensuring the vehicle stability and avoiding critical driving situations. Also, Fig. 11 shows a comparison between the

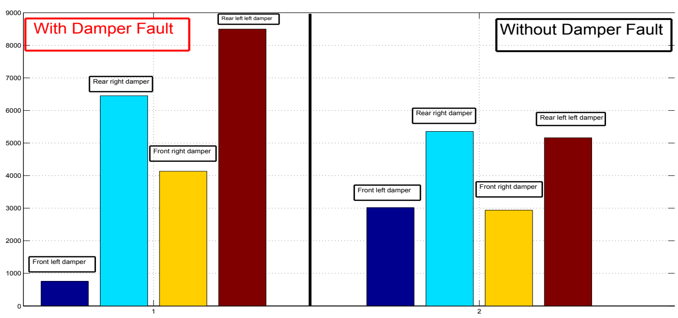

Fig. 11. RMS value of the suspension dampers: Faulty and Healthy case

RMS of the suspensions dampers forces in the faulty and healthy case. The proposed LPV/$\mathcal{H}_\infty$ suspension allocation control strategy allows in the healthy case to provide the accurate damping forces on each one of the vehicle four corners. Conversely, in the faulty case as presented on the left figure where a failure on the front left damper occurs, the proposed strategy reconfigures the suspension control by providing more damping forces on the other healthy dampers. This reconfiguration compensates the lack of the damping in the faulty front left corner and ensures the vehicle stability in the critical driving situations.

5. CONCLUSION

In this study, the efficiency of the new proposed LPV/$\mathcal{H}_\infty$ control reconfiguration structure to manage different actuators failures has been proved. This have led to a reliable fault tolerant control strategy that allows to prevent the risk of loss of manoeuvrability and safety degradation in critical driving conditions by using one of the important advantage of the LPV/$\mathcal{H}_\infty$ control that coordinates hierarchically the use of different actuators.

REFERENCES

Blanke, M., Izadi-Zamanabadi, R., Bøgh, S.A., and Lunau, C.P. (1997). Fault-tolerant control systemsa holistic view. *Control Engineering Practice*, 5(5), 693–702.

Do, A.L., Sename, O., and Dugard, L. (2010). An LPV control approach for semi-active suspension control with actuator constraints. In *Proceedings of the IEEE American Control Conference (ACC)*, 4653 – 4658. Baltimore, Maryland, USA.

Doumiati, M., Sename, O., Dugard, L., Martinez-Molina, J.J., Gaspar, P., and Szabo, Z. (2013). Integrated vehicle dynamics control via coordination of active front steering and rear braking. *European Journal of Control*, 19(2), 121 – 143.

Fergani, S., Sename, O., and Dugard, L. (2013a). A lpv suspension control with performance adaptation to roll behavior, embedded in a global vehicle dynamic control strategy. In *Control Conference (ECC), 2013 European*, 1487–1492.

Fergani, S., Sename, O., and Dugard, L. (2013b). A new LPV/$\mathcal{H}_\infty$ global chassis control through load transfer distribution and vehicle stability monitoring. In *Proceedings of IFAC Joint conference 5th Symposium on System Structure and Control, 11th Workshop on Time-Delay Systems 6th Workshop on Fractional Differentiation and Its Applications*. Grenoble, France.

Fergani, S., Sename, O., and Dugard, L. (2014). A lpv/$\mathcal{H}_\infty$ fault tolerant control of vehicle roll dynamics under semi-active damper malfunction. In *American Control Conference (ACC), 2014*, 4482–4487.

Gillespie, T. (1992). *Fundamental of vehicle dynamics*. Society of Automotive Engineers, Inc.

Kiencke, U. and Nielsen, L. (2000). *Automotive Control Systems*.

Milliken, W.F. and Milliken, D.L. (1995). *Race car vehicle dynamics*. Society of Automotive Engineers International.

Poussot-Vassal, C., Sename, O., Dugard, L., Gáspár, P., Szabó, Z., and Bokor, J. (2011a). Attitude and handling improvements through gain-scheduled suspensions and brakes control. *Control Engineering Practice*, 19(3), 252 – 263.

Poussot-Vassal, C., Sename, O., Dugard, L., Gaspar, P., Szabo, Z., and Bokor, J. (2011b). Attitude and handling improvements through gain-scheduled suspensions and brakes control. *Control Engineering Practice*, 19(3), 252–263.

Savaresi, S.M., Poussot-Vassal, C., Spelta, C., Sename, O., and Dugard, L. (2010). *Semi-active suspension control design for vehicles*. Access Online via Elsevier.

Scherer, C. (1996). Mixed $\mathcal{H}_2/\mathcal{H}_\infty$ control for time-varying and linear parametrically-varying systems. *International Journal of Robust and Nonlinear Control*, 6(9-10), 929–952.

Sename, O., Tudon-Martinez, J., and Fergani, S. (2013). Lpv methods for fault-tolerant vehicle dynamic control. In *Control and Fault-Tolerant Systems (SysTol), 2013 Conference on*, 116–130.